Numerical study of a hybrid absorption-compression high temperature heat pump for industrial waste heat recovery

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Abstract

The present paper aims at exploring a hybrid absorption-compression heat pump (HAC-HP) to upgrade and recover the industrial waste heat in the temperature range of 60~120 °C. The proposed new HAC-HP system has one condenser, one evaporator and one more solution pump comparing to the conventional HAC-HP system to allow flexible usage of energy sources of electricity and waste heat. In the proposed system, the pressure of ammonia-water vapor desorbed in the generator can be elevated by two routes, one is via the compression of compressor while the other one is via condenser, solution pump and evaporator. The results show that more ammonia-water vapor flow through compressor leads to substantial higher energy efficiency due to the higher quality of electricity, however, only slightly change on system exergy efficiency is noticed. The temperature lift increases with the increasing system recirculation flow ratio, however, meanwhile the system energy and exergy efficiencies drop towards zero. The suitable operation ranges of HAC-HP were recommended for the waste heat at 60 °C, 80 °C, 100 °C and 120 °C. The recirculation flow ratio should be lower than 9, 6, 5 and 4 respectively for these waste heat, while the temperature lifts were in the range of 9.8~27.7 °C, 14.9~44.1 °C, 24.4~64.1 °C and 40.7~85.7 °C respectively, and the system energy efficiency were 0.35~0.93, 0.32~0.90, 0.25~0.85 and 0.14~0.76.

Keywords: Absorption compression, high temperature heat pump, efficiency, industrial waste heat, thermodynamic analysis

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Nomenclature

FR	Recirculation flow ratio (-)
h	Enthalpy (J/kg)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (Pa)
$\dot{Q}$	Heat power (W)
R	Ratio (-)
T	Temperature (°C)
ΔT_{LMTD}	Logarithmic mean temperature difference (°C)
UA	Heat exchanger performance (W/K)
w	Mass fraction (-)
$\dot{W}$	Electric power (W)
η	Efficiency (-)
Subscripts	
abs	Absorption
amb	Ambient
bas	Basic
com	Compressor
en	Energy
eva	Evaporation
ex	Exergy
gen	Generator
H	High pressure
HE	Heat exchanger
l	Liquid

L	Low pressure
min	Minimum
pole	Pole
pump	Pump
rec	Rectifier
ref	Refrigerant
refl	Reflux
s	Isentropic
use	Useful
v	Vapor
was	Waste

1. Introduction

The industrial sector consumes about 54% of the total world's delivered energy [1], and the worse thing is that about one sixth of the total energy consumed by industrial sector is wasted as low-grade heat via radiation, exhausted gas, cooling fluid and so on [2]. This can be recovered by various technologies, such like power generation via organic Rankine cycle [3], refrigeration by absorption [4] and adsorption [5] technologies, and heat upgrade by heat pump [6], etc. Among these technologies, heat pump can efficiently pump the heat form low grade to high grade, and has been recognized as an efficient and practical solution to reduce greenhouse gas emission [7].

Comparing to conventional vapor-compression heat pump, the hybrid absorption-compression heat pump (HAC-HP) is believed to be possible to work for high temperature applications with a better system performance [8]. HAC-HP system is a combination between the conventional vapor-compression system and absorption system, the usage of binary mixture as working fluid leads to non-isothermal process of absorption/desorption and relatively lower temperature difference between the heat source and working fluid comparing to the isothermal condensation/evaporation process in conventional vapor-compression heat pump, thereby the system energy efficiency can be higher due to the lower cycle irreversibility and entropy generation. Some other benefits of using this hybrid system are listed below [9].

- Small swept volume of the compressor;
- Higher heat transfer coefficient;
- Environment-friendly refrigerant;
- More flexible in the changes of temperature level and capacity;
- Higher delivery temperature;

Brunin et al. [8] have compared the working domains of different heat pump technologies, and concluded that vapor-compression system using hydrocarbon fluid and hybrid system using ammonia-water were the only two technologies of high temperature heat pump as the abolishing of CFC and HCFC refrigerant. Minea and Chiriac [10] presented some design guidelines and operation experiences of ammonia-water based HAC-HP system, the 4.5 MW prototype has been tested in field and demonstrated the heat upgrade from 36 °C industrial cooling water to 55 °C useful hot water with a COP of 3.9. Kim et al. [11] experimentally tested a 10 kW HAC-HP system using ammonia-water as working fluid, and elevated the heat from 50 °C to over 90 °C with ammonia mass fraction in the weak solution of 0.42. Jensen et al. [12] numerically investigated the influences of ammonia mass fraction and liquid circulation ratio on the system constrains, results showed that the maximum heat supply temperature was 111 °C when using standard refrigeration components without modifying the compressor (28 bar limitation); for high pressure ammonia based components, the maximum supply temperature can be 129 °C (50 bar limitation) and for transcritical CO_2 based components, the maximum supply temperature can be 147 °C (140 bar limitation). The authors [13] also thermodynamically analysed HAC-HP system to recover waste heat from spray-drying facilities, exergoeconomic analysis was conducted to minimize the lifetime cost. The best solution they reported was an 895 kW heat pump with ammonia mass fraction at 0.82 and a circulation ratio of 0.43. This system can generate economic saving of €146,000 and annual CO_2 emission reduction of 227 ton. Bourouis et al. [14] studied a single stage HAC-HP system using ternary mixture of

Trifluoroethanol-water-tetraethylenglycol dimethylether (TFE-H_2O-TEGDME) as working fluid, the results showed that the system can upgrade the thermal waste heat form 80 °C to 120 °C at a COP of 6.4.

The conventional HAC-HP systems mentioned above are all based on a similar configuration with main components of generator, absorber, solution pump and compressor. Such system uses electricity as the main driven force to upgrade the low-grade waste heat, though the system COP is high, the operation cost of conventional HAC-HP system is still proportionally increased with the amount of recovered waste heat. The current paper proposed adding one condenser, one evaporator and one more solution pump to the system to form a HAC-HP system using compressor and condenser-pump-evaporator together to elevate the pressure of the working fluid. There are two benefits of employing such dual energy source system. First, the system can be more flexible on the usage of different energy source, i.e. electricity and/or waste heat, depending on their availabilities; secondly, more waste heat can be recovered as the system required thermal inputs of both generator and evaporator while the required electricity can be less. The energy efficiency, exergy efficiency, temperature lift and useful heat power were numerically calculated and analysed.

2. Working principle and analysis method

Ammonia-water is used as working fluid in current study. The correlations developed by El-Sayed and Tribus [15] were used to determine the equilibrium liquid and vapor states of ammonia-water mixture; while Gibbs free energy formulations reported by Ziegler and Trepp [16] were used to calculate the enthalpies and entropies of ammonia-water mixture. The following hypothesises were used to simplify the numerical analysis.

- The cycle was operated at steady-state;
- The solutions at the outlet of condenser and evaporator were at saturated state;
- The solutions at the outlet of generator and absorber were at saturated state;
- The vapor from the rectifier was at saturated state;
- Throttling doesn’t change the solution enthalpy;

- After throttling, the solution entering the generator was at two-phase or saturated liquid state;
- Pressure drop and heat loss in the system were both negligible.

The schematic diagram of the studied HAC-HP system is shown in Figure 1. The system consists of a column shape generator, a rectifier on the top of the generator, a recuperator, an absorber, a condenser, an evaporator, a compressor, two solution pumps and a throttling valve cooling

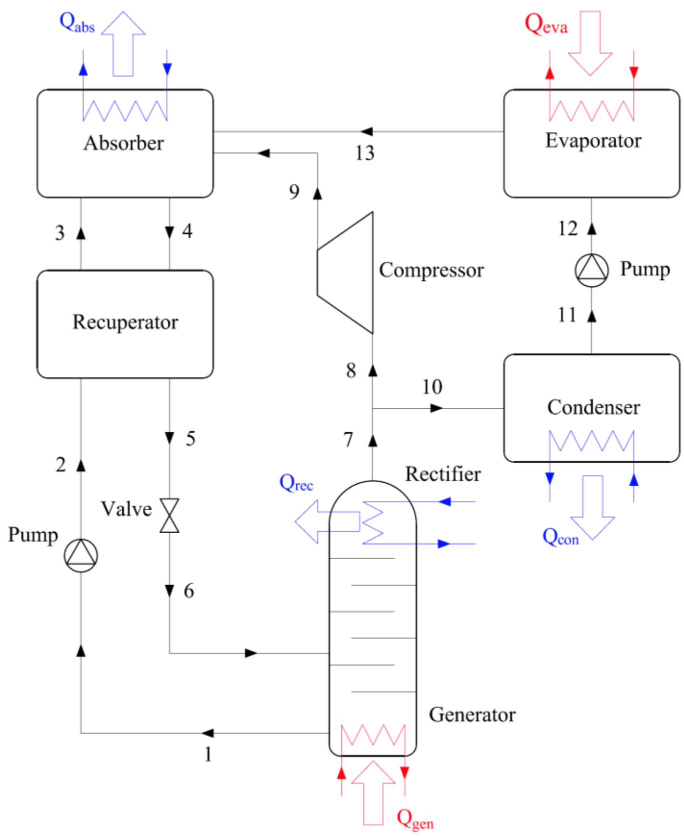

Figure 1 Schematic diagram of hybrid absorption-compression heat pump.

There are two pressure levels in the whole cycle, low pressure P_L in the generator, rectifier and condenser, and high pressure P_H in evaporator, absorber and recuperator. In current study, the refrigerant ammonia mass fraction (after the rectifier), w_{ref}, is pre-defined at 0.9995 as suggested by [17]. According to the predefined assumption, the ammonia-rich vapor at the outlet of rectifier is at saturated state, so that

P_L can be determined as the saturated vapor pressure at rectification temperature, T_{rec} and ammonia mass fraction w_{ref}. Meanwhile, P_H can be determined by the saturated vapor at the outlet of evaporator based on the waste heat temperature T_{was} and w_{ref}. On the basis of P_L and w_{ref}, the condensation temperature, T_{con}, can be determined based on the saturated liquid state at the outlet of condenser. This condensation temperature should be slightly lower than the rectification temperature due to the temperature difference between saturated liquid and vapor of ammonia-water at the same pressure and ammonia mass fraction. Then other thermodynamic states of saturated liquid at the outlet of condenser, saturated vapors at the outlet of evaporator and rectifier can be determined.

The ammonia-water liquid at the outlet of generator is at saturated state, where $T_1 = T_{was}$ and $P_1 = P_L$, hence the ammonia mass fraction of the used basic ammonia-water liquid (weak solution), w_{bas}, can be determined by the thermodynamic state equation. Thereafter, the enthalpy h_1 and entropy s_1 can be calculated. The recirculation flow ratio (FR) [17] is used to calculate the mass flow rate of the ammonia-water liquid, the definition of FR is in Eq. (1).

$$FR = \frac{\text{mass flow rate of liquid leaving generator}}{\text{mass flow rate of vapor leaving generator}} = \frac{\dot{m}_1}{\dot{m}_7} \quad (1)$$

Then mass flow rate of the refrigerant, $\dot{m}_7$, was pre-defined at 0.01 kg/s to allow an approximately 12 kW (refrigeration power) system according to the evaporation enthalpy change of ammonia. Thereafter, the ammonia-water liquid mass flow rates $\dot{m}_1$, $\dot{m}_6$ and the ammonia-rich liquid mass fraction w_6 can be calculated based on the mass balance equations. The temperature of the useful heat, $T_{use} = T_4$, is then determined by the saturated liquid at the outlet of absorber at the pressure of P_H and the mass fraction of pump w_4 ($w_4 = w_6$); also the enthalpy h_4 can be calculated.

The condensed ammonia-water liquid is pumped from P_L to P_H, the isentropic efficiency, η_{pump1}, of this solution pump is given as

$$\eta_{pump1} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (2)$$

where h_{2s} is the enthalpy of ammonia-water liquid at the outlet of pump if the process is isentropic. The η_{pump1} has been pre-defined at 0.85 according to the literature [18]. h_{2s} can be calculated by considering the ammonia-water liquid at pressure P_H with an entropy value of s_1, then the value of h_2 can be calculated by Eq. (2). The consumed pumping power is then calculated by the following equation.

$$\dot{W}_{\text{pump1}} = \dot{m}_1(h_2 - h_1) \tag{3}$$

The same method can be applied to the second pump located between condenser and evaporator to obtain $\dot{W}_{\text{pump2}}$. Meanwhile, the following two equations can be employed to solve the compression process using the similar procedure as for the pump.

$$\eta_{\text{com}} = \frac{h_{9s} - h_8}{h_9 - h_8} \tag{4}$$

$$\dot{W}_{\text{com}} = \dot{m}_8(h_9 - h_8) \tag{5}$$

where η_{com} has been pre-defined at 0.75 [18], the mass flow rate of ammonia-water vapor entering the compressor can be calculated based on the value of splitting ratio, $R_{\text{ref}} = \frac{\dot{m}_8}{\dot{m}_7}$.

For the recuperator, thermal states of the two inlet ammonia-water liquids (T_2, h_2, T_4, h_4) have been determined, then the outlet temperature (T_3 and T_5) and enthalpy (h_3 and h_5) of the two liquids can be iteratively determined through heat balance equations, Eq. (6) and Eq. (7), and the logarithmic temperature difference ΔT_{LMTD} in Eq. (8).

$$\dot{Q}_{\text{HE}} = \Delta T_{\text{LMTD}} \cdot UA \tag{6}$$

$$\dot{Q}_{\text{HE}} = \dot{m}_2(h_3 - h_2) = \dot{m}_4(h_4 - h_5) \tag{7}$$

$$\Delta T_{\text{LMTD}} = \frac{T_4 - T_3 - (T_5 - T_2)}{\ln\left(\frac{T_4 - T_3}{T_5 - T_2}\right)} \tag{8}$$

The rectification reflux ratio, R_{refl}, can be used to assist the calculation of the rectification process [19]. As shown in Figure 2, the limited height of the rectification column leads to a steeper operation line comparing to the ideal isothermal rectification line. The reflux ratio is defined as

$$R_{\text{refl}} = \frac{h_{\text{pole}} - h_7}{h_{\text{min}} - h_7} \tag{9}$$

where h_{min} can be calculated by

$$h_{\text{min}} = h_{6\text{v}} + (h_{6\text{v}} - h_{6l})\frac{w_{\text{ref}} - w_{6\text{v}}}{w_{6\text{v}} - w_{6l}} \tag{10}$$

R_{refl} of 2 was used in current study according to the recommendation by the literature [19], then h_{pole} can be calculated by Eq. (9). The rectification heat is then

$$\dot{Q}_{\text{rec}} = \dot{m}_7 \left(h_{\text{pole}} - h_7 \right) \tag{11}$$

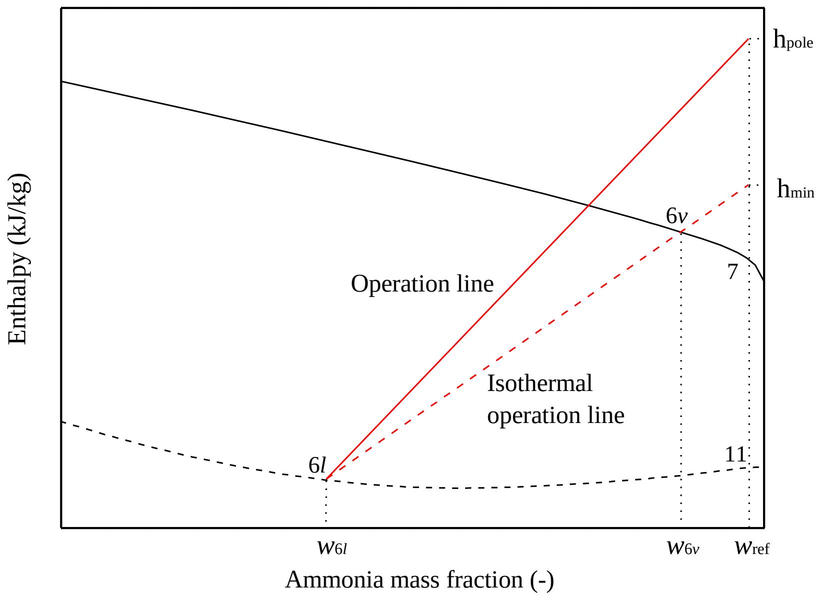

Figure 2 Determination of the pole of rectification.

The generation, evaporation and absorption heat can be calculated by the following equations.

$$\dot{Q}_{\text{gen}} = \dot{m}_1 h_1 + \dot{m}_7 h_7 + \dot{Q}_{\text{rec}} - \dot{m}_6 h_6 \tag{12}$$

$$\dot{Q}_{\text{eva}} = \dot{m}_{12} (h_{13} - h_{12}) \tag{13}$$

$$\dot{Q}_{\text{abs}} = \dot{m}_3 h_3 + \dot{m}_9 h_9 + \dot{m}_{13} h_{13} - \dot{m}_4 h_4 \tag{14}$$

Finally, the energy efficiency and exergy efficiency of the HAC-HP cycle can be calculated by Eq. (15) and Eq. (16), respectively.

$$\eta_{\text{en}} = \frac{\dot{Q}_{\text{abs}}}{\dot{Q}_{\text{gen}} + \dot{Q}_{\text{eva}} + \dot{W}_{\text{com}} + \dot{W}_{\text{pump1}} + \dot{W}_{\text{pump2}}} \tag{15}$$

$$\eta_{\text{ex}} = \frac{\dot{Q}_{\text{abs}} (1 - T_{\text{amb}} / T_{\text{use}})}{(\dot{Q}_{\text{gen}} + \dot{Q}_{\text{eva}})(1 - T_{\text{amb}} / T_{\text{was}}) + \dot{W}_{\text{com}} + \dot{W}_{\text{pump1}} + \dot{W}_{\text{pump2}}} \tag{16}$$

The used parameters in the calculation are summarized in Table 1.

Table 1 Parameters used in the calculation.

Parameter	Value
T_{was} (°C)	60-120
FR (-)	1-20*
R_{ref} (-)	0-1
R_{refl} (-)	2
η_{pump1}, η_{pump2} (-)	0.85
η_{com} (-)	0.75
UA (W/K)	1000
w_{ref} (-)	0.9995
$\dot{m}_{ref}$ (kg/s)	0.01

* *FR* was initially set from 1 to 20, however, the feasible values of *FR* were limited by each working condition

3. Results and discussion

Figure 3 shows an example of the operation states of the HAC-HP system in the enthalpy-mass fraction chart, the waste heat temperature is 80 °C and FR = 7. The two pressure levels in the system are 11.1 bar and 38.9 bar while the ammonia-lean and ammonia-rich solution have the mass fraction at 0.412 and 0.485 respectively, these give the pre-defined refrigerant mass fraction at 0.9995.

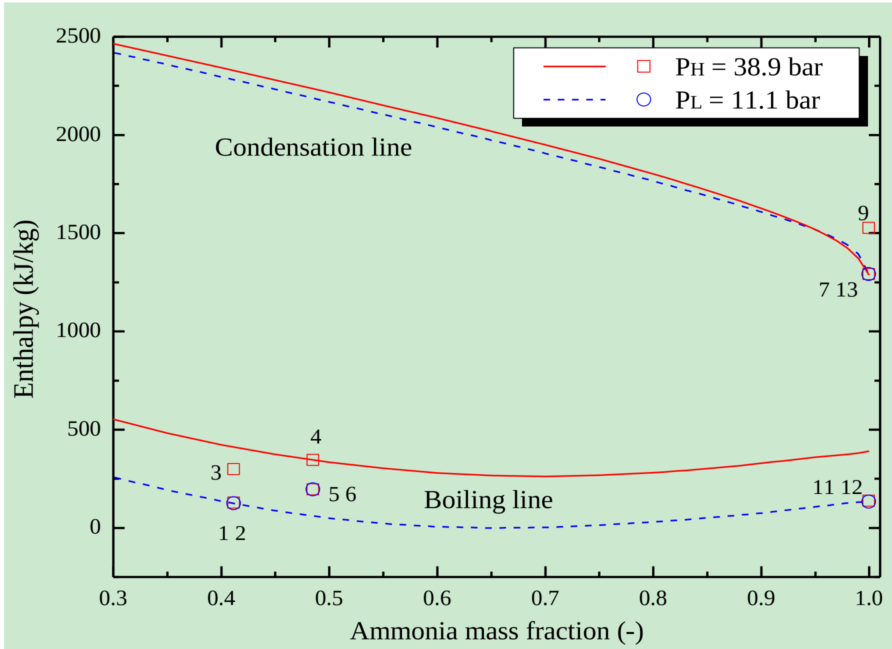

Figure 3 Thermodynamic states of the working fluid at different points, T_{was} = 80 °C, *FR* = 7.

Figure 4 shows the energy efficiency of the proposed HAC-HP system with waste heat temperature at 80 °C as an example. The maximum energy efficiency obtained in the studied working conditions is approximately 0.90 as shown in the figure. This maximum value occurs at the condition of *FR* = 1 and R_{ref} = 0. These curves indicate that using compressor is more efficient than using condenser-pump-evaporator as can be seen that lower refrigerant splitting ratio leading to higher energy efficiency. For example, the highest η_{en} is about 0.90 when using compressor only at the condition of *FR* = 1; while the highest η_{en} is only about 0.43 if using condenser-pump-evaporator only. The upgraded temperature is from 95 °C to approximately 132 °C, which gives a temperature lift in the range of 15–52 °C. Higher *FR* leads to smaller ammonia mass fraction difference between ammonia-rich and ammonia-lean solution, then the saturated ammonia-water liquid from absorber has relatively lower ammonia mass fraction, therefore the equilibrium temperature in the absorber at a certain pressure (P_H has been determined by evaporator) is higher. A relatively flat energy efficiency curve can be noticed when *FR* is smaller than 6 for all values of R_{ref}, the change of energy efficiency is no more than 25%; then the energy efficiencies drop dramatically towards zero as *FR* is higher than 7.

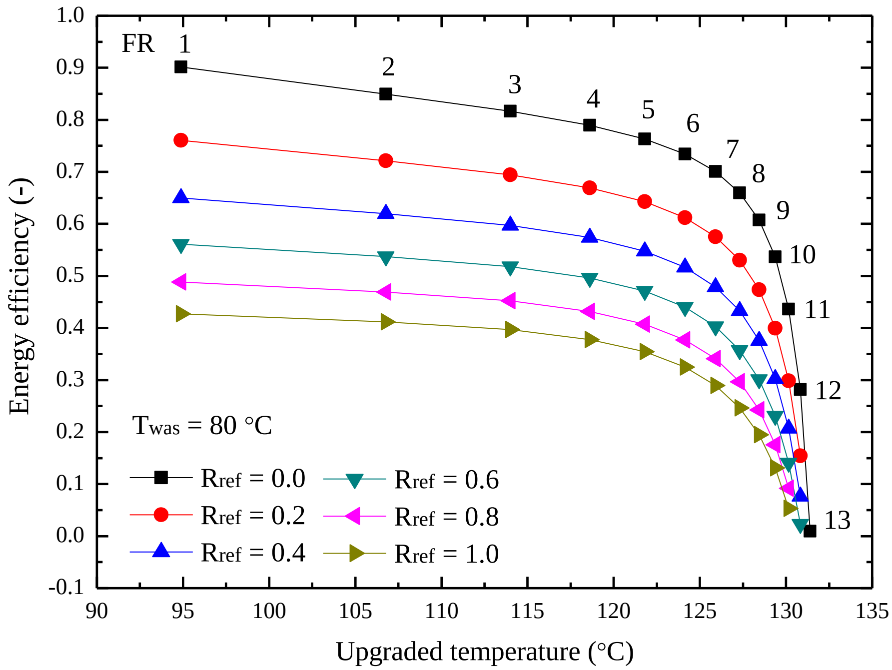

Figure 4 Energy efficiency of HAC-HP with waste heat temperature at 80 °C.

Figure 5 shows the exergy efficiency of the system with 80 °C waste heat. Comparing to the energy efficiency curves in Figure 4, the gap between each exergy efficiency curve having different values of R_{ref} is smaller, e.g. less than 8%. The exergy efficiency increases gently as *FR* increasing until 3, followed by decreasing gently until 6 and reducing violently towards zero. Based on these values of exergy efficiency, the optimal operating value of *FR* locates in the range of 1~6, where the exergy efficiency varies in the range of 0.50~0.61 while the corresponding energy efficiency is 0.33~0.90 and the temperature lift is 14.9~44.1 °C.

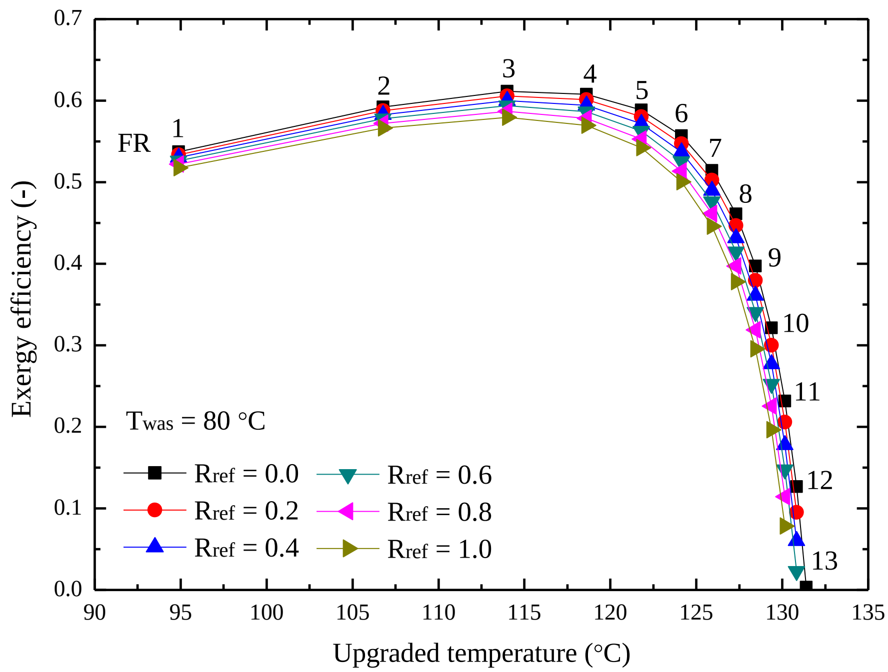

Figure 5 Exergy efficiency of HAC-HP with waste heat temperature at 80 °C.

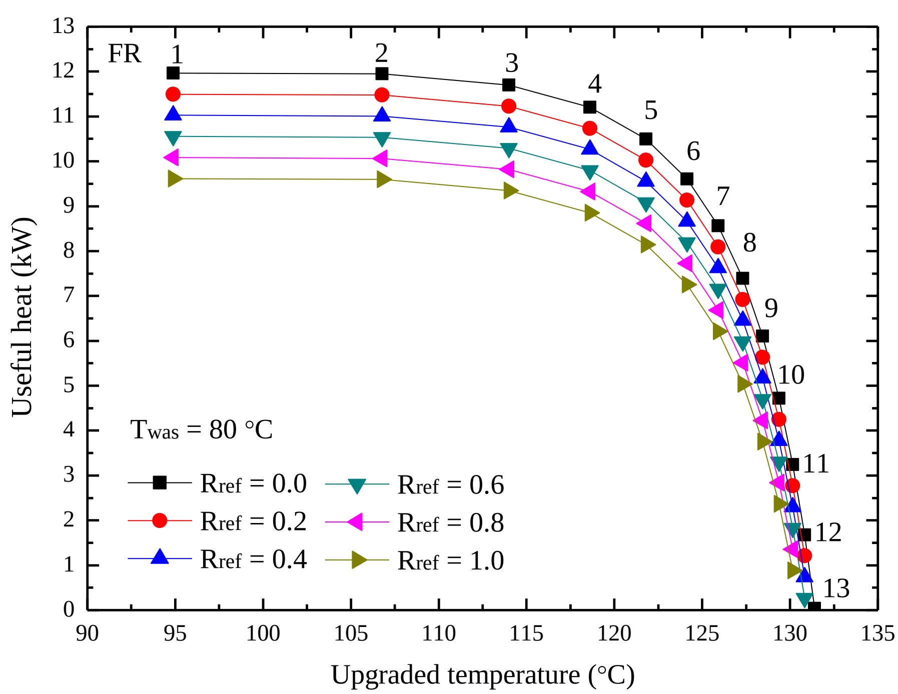

Figure 6 Useful heat of HAC-HP with waste heat temperature at 80 °C.

Figure 6 shows the upgraded useful heat power with 80 °C waste heat. The power of useful heat is only slightly reduced as *FR* increasing from 1 to 4, e.g. from 11.97 kW to 11.20 kW in the case of R_{ref} = 0 and from 9.6 kW to 8.9 kW in the case of R_{ref} = 1. Thereafter the useful heat power drops significantly towards zero as shown in the figure. To obtain these amount of useful heat, the required input heat and electricity power are shown in Figure 7. The required electricity consumed by compressor and solution pumps is less than 3 kW, while the required heat for the generator and evaporator can be as high as nearly 24 kW. As shown in the figure, when R_{ref} = 0 more electricity and less heat are consumed comparing to that in the case of R_{ref} = 1, however, the extent of variation of heat is more notable than that of electricity due to the lower exergy possessed by thermal energy than electricity. More heat consumption indicates lower energy efficiency. Nevertheless, if treating the waste heat as totally free energy and only electricity is counted in Eq. (15), the system energy efficiency (or coefficient of performance) will be significantly higher and increase with the drop of R_{ref}.

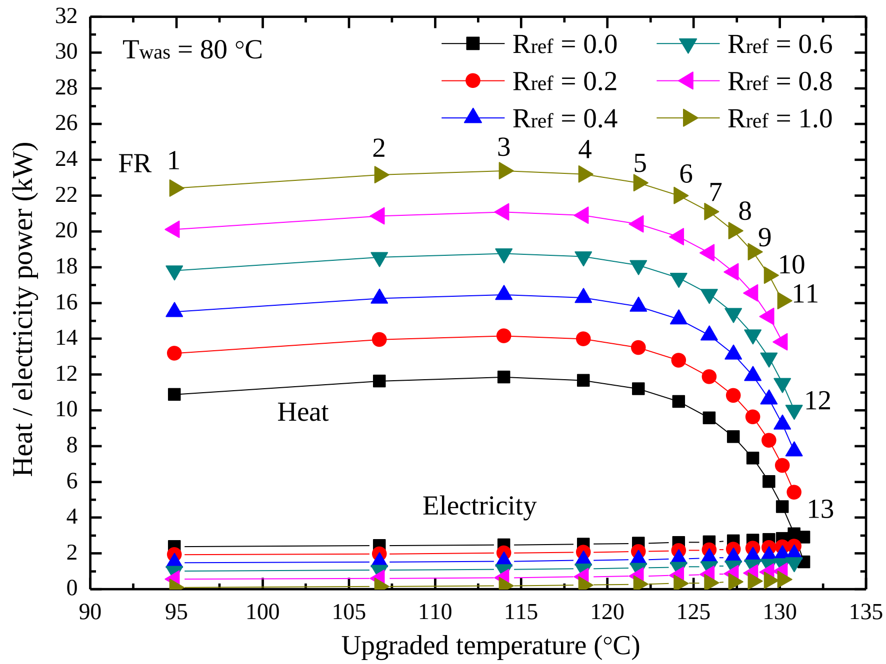

Figure 7 Input heat and electricity power with waste heat temperature at 80 °C.

Figure 8 shows the energy efficiencies of HAC-HP system as the function of temperature lift at different waste heat temperatures from 60 °C to 120 °C. As shown in the figure, the energy efficiency curves shift to lower right side as the increase of waste heat temperature, towards higher temperature lift but lower energy efficiency. For example, for 120 °C waste heat, the temperature lift is in the range of 40.7~101.0 °C with the energy efficiency lower than 0.76 with compressor only and lower than 0.28 with condenser-pump-evaporator only. Table 2 presents the recommended operation conditions and corresponding performance of HAC-HP system before the violent drop of system energy and exergy efficiencies and output useful heat power.

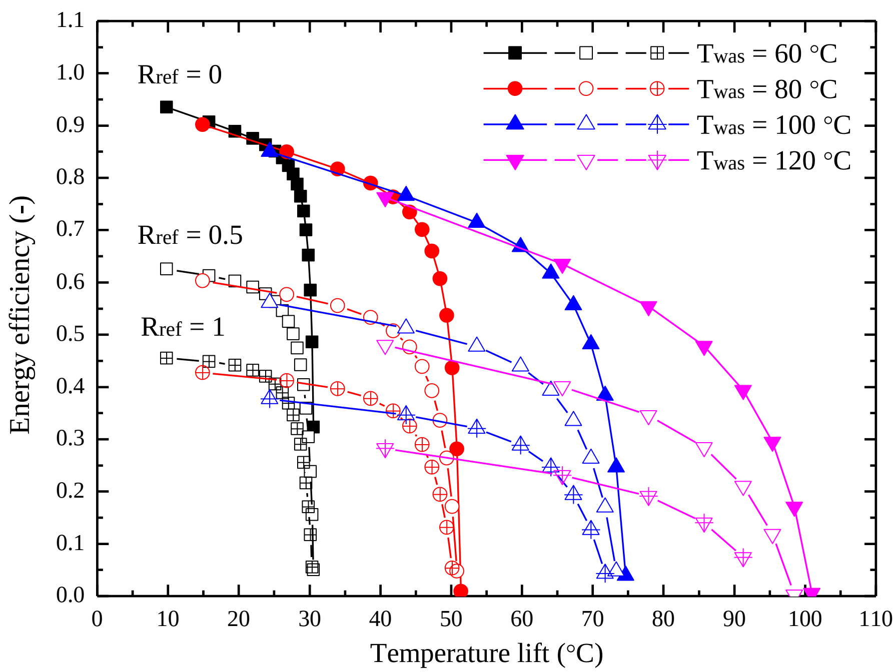

Figure 8 Energy *vs* temperature lift of HAC-HP with waste heat temperature of 60~120 °C.

Table 2 Recommended *FR* and corresponding performance of HAC-HP system at different waste heat temperature, R_{ref} = 0~1.

Waste heat temperature	Recommended *FR* (-)	Energy efficiency (-)	Exergy efficiency (-)	Useful heat (kW)	Temperature lift (°C)

(°C)					
60	< 9	0.35~0.93	0.55~0.67	7.2~11.7	9.8~27.7
80	< 6	0.32~0.90	0.50~0.61	7.3~11.9	14.9~44.1
100	< 5	0.25~0.85	0.37~0.56	5.7~11.9	24.4~64.1
120	< 4	0.14~0.76	0.21~0.50	3.5~11.2	40.7~85.7

4. Conclusions

The present study investigated a hybrid absorption-compression heat pump to upgrade and recover the industrial waste heat in the temperature range of 60~120 °C. The proposed system can use two routes to elevate the working fluid pressure, one is via compressor and the other is via condenser-pump-evaporator, so that the usage of energy sources of electricity and waste heat can be flexible. The major conclusions are summarised as follows.

(1) As the increase of system recirculation flow ratio, the temperature lift was improved; however, the system energy and exergy efficiencies dropped violently after the recirculation flow ratio larger than certain value;

(2) More ammonia-water vapor flowing through compressor led to higher system energy efficiency; however, the exergy efficiency changed little with the vapor splitting ratio;

(3) The recommended recirculation flow ratio should be lower than 9, 6, 5 and 4 respectively for waste heat at 60 °C, 80 °C, 100 °C and 120 °C respectively, while the temperature lifts were 9.8~27.7 °C, 14.9~44.1 °C, 24.4~64.1 °C and 40.7~85.7 °C respectively, and the system energy efficiency were 0.35~0.93, 0.32~0.90, 0.25~0.85 and 0.14~0.76.

Acknowledgement

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